Comparison of Transmission Error Predictions With Noise Measurements for Several Spur and Helical Gears

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Abstract

Measured sound power data from eight different spur, single and double helical gear designs are compared with predictions of transmission error by the Load Distribution Program. The sound power data was taken from the recent Army-funded Advanced Rotorcraft Transmission project. Tests were conducted in the NASA gear noise rig. Results of both test data and transmission error predictions are made for each harmonic of mesh frequency at several operating conditions. In general, the transmission error predictions compare favorably with the measured noise levels.

Introduction

There has long been a strong feeling that transmission error is closely related to the noise emitted by gear pairs [1-7]. However, few direct comparisons of transmission error predictions with actual measured sound data exist. Although transmission error predictions have been made by several investigators [5-7], there have been few published efforts that attempt to corroborate predictions with systematically designed experimental tests. The most convincing of the comparisons that has been performed is that of Kubo [11], who, after performing some adjustments to account for manufacturing variations from tooth to tooth, shows a good correlation between noise and predicted transmission error for several similar geometry gear sets. The literature [8-11] shows that many people have investigated the effects of different gear geometries on noise, but few have made controlled measurement of different geometries in one housing.

In a study to create a data base for gear noise assessment, Drago et al. [12] designed and made gears of eight different geometries. The gears were tested in a NASA-Lewis gear noise test rig. A logical follow-up of the acquisition of test data is the comparison of the data to transmission error models. This paper uses a computer model called the Load Distribution Program (LDP), a program that predicts transmission error for all types of parallel axis gears [7], to model the gears and to compare the predictions to the measured results. The results presented show a good correlation between the predictions and measured data.

Test Data

The test gears were driven by an electric motor and loaded with an electric dynamometer. The test gears were mounted on shafts in a 3.5 inch center distance housing. Some power measurements reported in [12] were taken with an acoustic intensity probe scanned over a prescribed measurement grid just above the top of the gear box. Reported sound power measurements were the sum of the amplitudes of the frequency peaks at the first three mesh harmonics and the amplitudes of the significant sidebands around each mesh harmonic. Further details of the test rig and measurement procedures are reported in [12].

The basic gear geometry is given in Table 1 and the 8 gear pairs tested are summarized in Table 2. The profile contact ratios and face contact ratios, which differ slightly from those given by Drago, et al. [12], were computed by using tooth tip roll angles taken from profile inspection charts. Henceforth, gear pairs will be referred to with the numbers given in the first column of Table 2 and the letters following each number will indicate the torque loading (an A indicates the lowest torque of 1361 lbf-in. and a C indicates the highest torque of 2269 lbf-in.).

Test Gears and Test Parameters

The test gear pairs chosen in Table 2. provide a broad range of geometries including a baseline spur gear pair, a

Table 1. Test Gear Configuration

	Pinion	Gear
Number of Teeth	25	31
Diametral Pitch, Transverse	8.00	
Center Distance	3.50	
Pressure Angle, Transverse	25 (Std Profile Contact Ratio) 20 (High Profile Contact Ratio)	
Face Width (Spur & Single Helical)	1.25	
Face Width (Double Helical)	Double Helicals 0.625 ea Helix	

Table 2. Geometry of Test Gear Sets

Configuration	Tooth Form	Type	Contact Ratios		
			Profile	Face	Total
1. Conventional Spur Baseline	Involute	Spur	1.43	0.00	1.43
2. HCR-INV	Involute	Spur	2.10	0.00	2.10
3. Conventional Single Helical Baseline	Involute	Helical	1.38	1.25	2.63
4. Double Helical	Involute	Helical	1.41	2.27	3.68
5. Helical-INV	Involute	Helical	1.37	1.76	3.13
6. HCR Helical-INV	Involute	Helical	2.12	2.25	4.37
7. NIF Baseline	NonInvolute	Spur	1.43	0.00	1.43
8. NIF-HC	NonInvolute	Spur	2.20	0.00	2.20

high contact ratio spur gear pair and several helical gear geometries that includes a double helical gear pair. The gears were ground to an aircraft quality (AGMA class 12) with virtually straight leads (no crowning). In most of the gear pairs, both the gear and the pinion had both tip and root relief that started at the pitch and tapered fairly linearly to the tip and root, respectively (see Fig. 4). Except for the double helical pair, the sets have the same inertias and the same mounting configuration. The width of the double helical set is somewhat greater than the others due to the gap between the two helices and the bearings used to support the double helicals were different than the other sets.

Gear pairs 7 and 8 are two heavily modified sets that have the purported advantage of having a more constant radius of curvature, thus enhancing both surface durability and scoring resistance.

Tests were run at three speeds (3000, 4000 and 5000 rpm) and three torques (1361, 1816, and 2269 lbf-in.). The middle load almost always had response levels in between the two extreme loads, both in the experiment and in the LDP simulation, so it is felt that just presenting the extreme loads provides a good feel for the results.

Noise Results

The Drago, et al.[12] presentation of noise results used the sum of the amplitudes of the first three gear mesh harmonics as well as sidebands about these harmonics to compute overall sound power levels in dB. However, in reviewing the harmonic data it was observed that each harmonic was significantly affected by gear test rig dynamics, since at some speeds the mesh frequency dominated and at other speeds one of the higher harmonics dominated the sound power level. Likewise, the LDP predictions provide information on each harmonic, so it is logical that the experimental and predicted data are presented one harmonic at a time.

Therefore, Figs. 1-3 summarize the experimentally generated data base for each of the respective harmonics of mesh frequency. The ten bar charts for each test condition include sound power data from three separate test runs of the "baseline" spur gear pair (gear pair number #1) and data for

each of the other test gear pairs given in Table 2. A tremendous amount of data is presented in these three figures, however, a cursory scan of them shows several trends, some of which have been pointed out by Drago et al. [12]:

1. The statistical variation in sound powers of the three "repeat" runs of the reference gear pair is usually about 2-4 dB.

2. At each test condition, the trends from one gear pair to the next for each harmonic are quite similar, i.e. in most cases the non-involute gears numbered 7 and 8 have the highest sound power levels, the spur gear pair #1 is next highest, and the helical gears are consistently the quietest.

3. With the exception of gear pairs 7 and 8, there is a definite correlation between sound power level and total contact ratio for each of the harmonics of mesh frequency.

4. In every case, gear pair 6, the high profile/high face contact ratio helical set, has the lowest sound power level.

5. There is a strong speed effect, as indicated by the relative amplitudes of the harmonics at each speed. For instance, the third harmonic has the highest levels at 3000 rpm, the second harmonic is highest at 4000 rpm and the fundamental harmonic is highest at 5000 rpm. Although not shown, a three-dimensional "waterfall" plot verified that these "high" levels at harmonics are indeed influenced by system resonances. It is interesting to note that the first harmonic amplitudes shifted by as much as 25 dB as speed was changed. These wide shifts in measured sound power amplitudes are due to the dynamics of some part of the shafting/housing system that are certainly important in ascertaining the noise radiation. These dynamics are not being evaluated in this paper, but by considering one operating condition at a time when comparing the performance of the gear sets, these dynamic effects are essentially "normalized" out.

Load Distribution Program Modeling of Profile Modifications

The Load Distribution Program uses a simplex type algorithm [13] to compute the load distribution along the lines of contact of spur and helical gears. The prediction of transmission error of the gear set is a byproduct of the load distribution computation. Also computed in the program are root stresses [14], time varying bending moments, and bearing forces. The deflections accounted for in the computation include tooth bending and shear using a Rayleigh Ritz procedure[15,16], Hertzian deflection, tooth base rotation and base translation [17], shaft deflections, and bearing and housing deflections. Intentional deviations from perfect teeth, including profile and lead modifications are accounted for, as are misalignment errors and tooth manufacturing errors that include. pressure angle errors, spacing errors, and lead errors. Errors and modifications may be entered as combinations of straight lines, parabolas, and/or circular forms or may be entered from digitized profile and/or lead checks. Multi-variable analysis is possible [18] so that
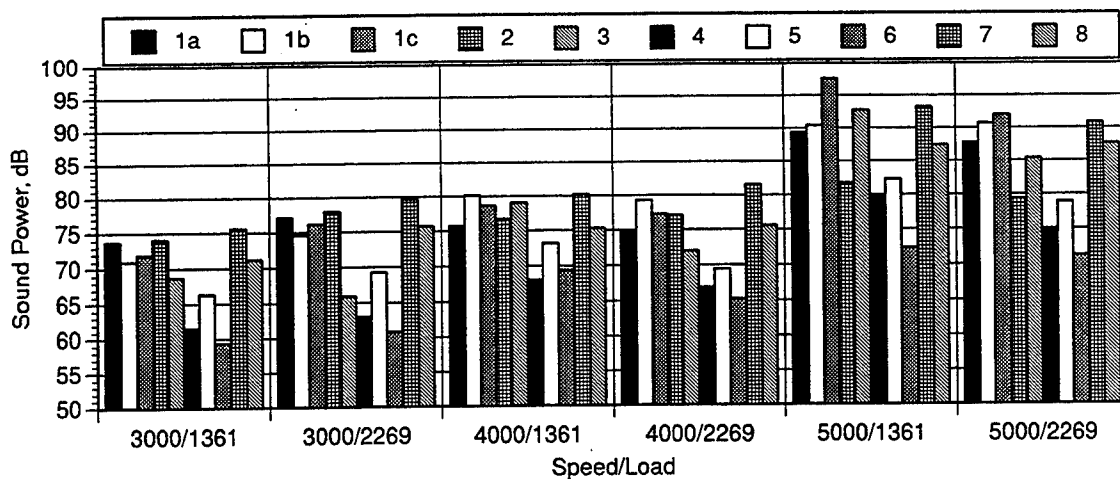
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Fig. 1.   Summary of the Sound Power at Mesh Frequency (Including Two Adjacent Sideband Pairs) for Each of the Test Gear Pairs
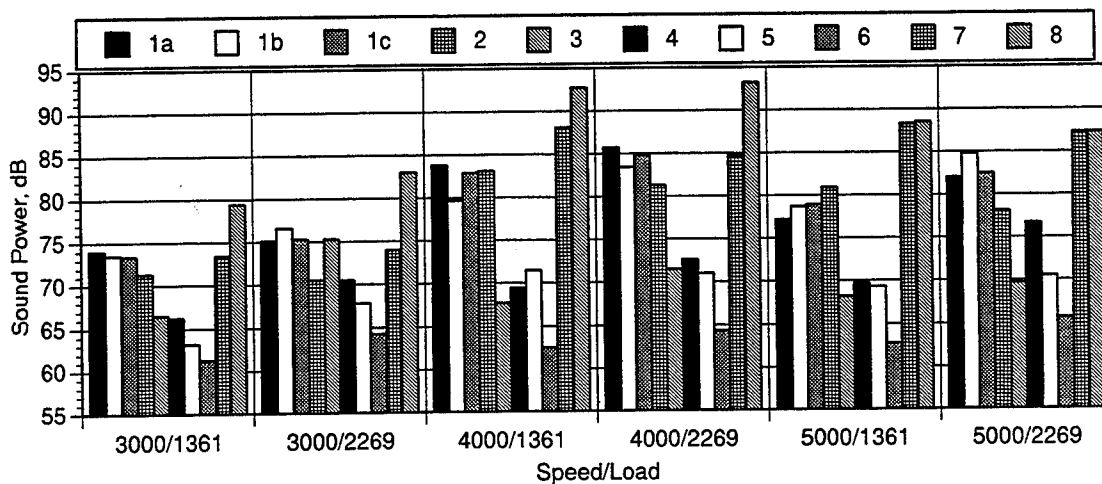

Fig. 2.   Summary of the Sound Power at Twice Mesh Frequency (Including Two Adjacent Sideband Pairs) for Each of the Test Gear Pairs
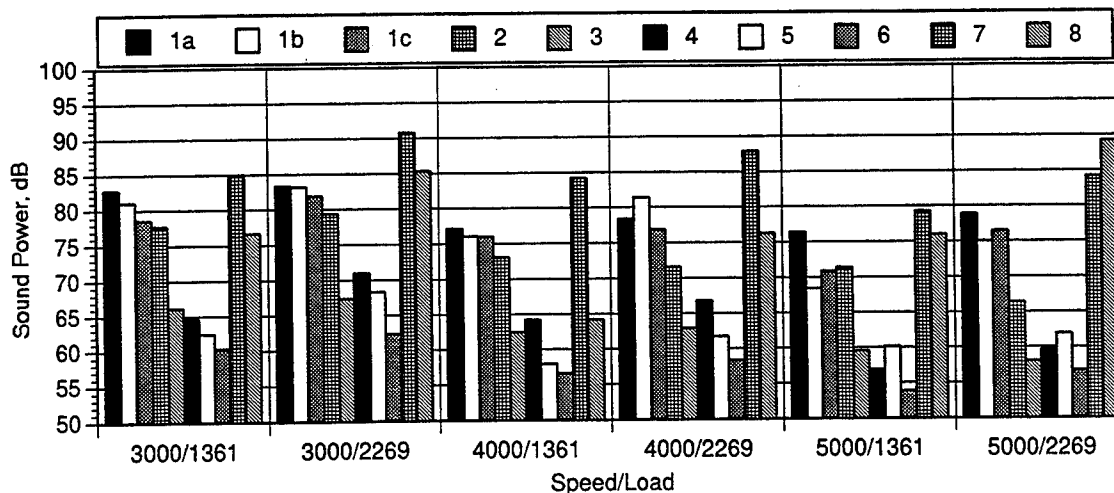

Fig. 3.   Summary of the Sound Power at T hree Times Mesh Frequency (Including Two Adjacent Sideband Pairs) for Each of the Test Gear Pairs

manufacturing sensitivities are easily studied. Several of these features will be shown in the analysis of the test gears.

It has long been understood that profile modifications, especially for spur gears, provide certain advantages in designing quiet gears [1,19,20] It has been shown that conventional tip and/or root relief give a minimum noise at only one operating load [7,20,21]. Fig. 4 shows the profile modifications for the baseline spur gear and pinion, respectively (gear pair 1). These charts are fairly typical of those of gear pairs 1-6, with the amplitudes and shapes changing a small amount for each gear pair. Although LDP accepts the digitized input of profile charts, it was found that each of the inspection charts could be emulated in LDP with combinations of straight lines and parabolas. Using these approximations also allows for some eyeball averaging of the individual profile charts.
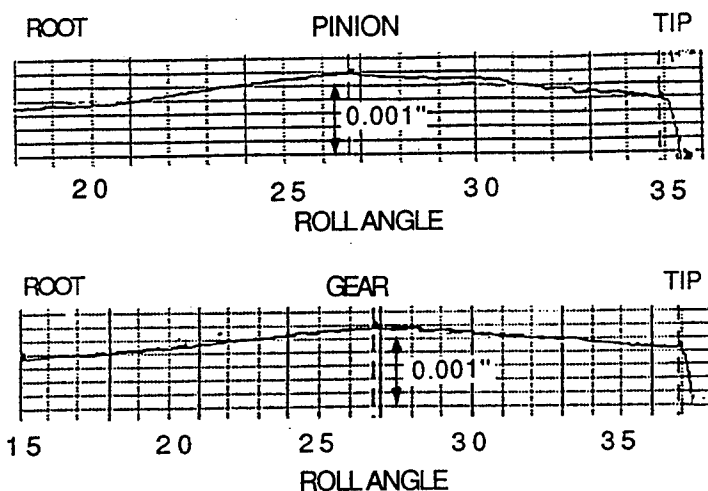

Fig. 4. Measured Profiles of the Baseline Spur Gear Pair

Fig. 5 shows the profiles created in LDP to simulate the profile charts of Fig. 4.. The last curve in Fig. 5 provides the sum of the modifications of the pinion and gear when added at their corresponding roll angles for each contact position. This last figure is the "effective" tooth pair modification.

As an example of the LDP analysis, both the baseline spur gear pair and the high contact ratio spur gear pair (Pair #2) will be used to show typical LDP results. 1. Fig. 6 shows a "poor person's" load contour map for the 11 contact positions of the baseline spur gear set (gear pair #1). In this Figure, load intensity is normalized such that the higher loads have higher numbers so that a "zero" indicates no contact and a nine is a the highest load intensity. Here, contact extends across the entire profile (no zeros) and tapers towards zero at the tips, indicating that the modification was selected quite well The predicted transmission error trace over one mesh cycle and the amplitudes of the first three mesh frequency harmonics are presented in Fig. 7.

When the amplitude of the mesh frequency harmonic of transmission error is plotted versus torque of the spur gear
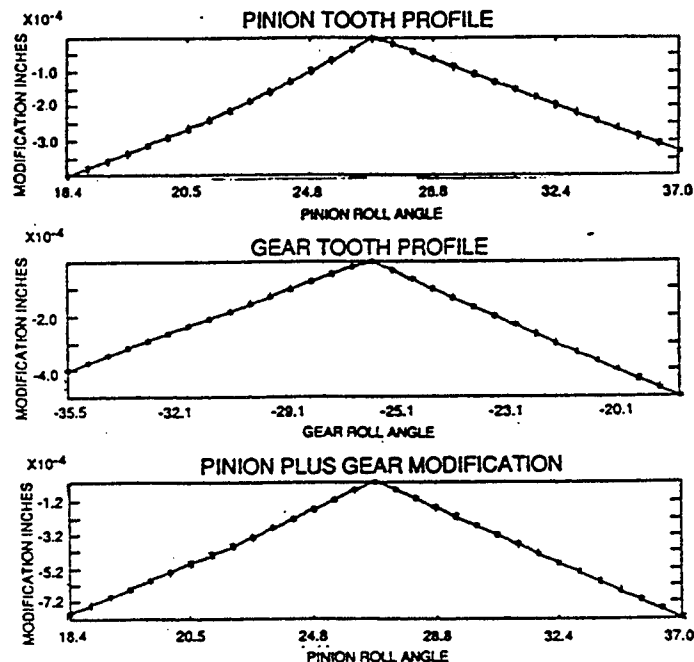

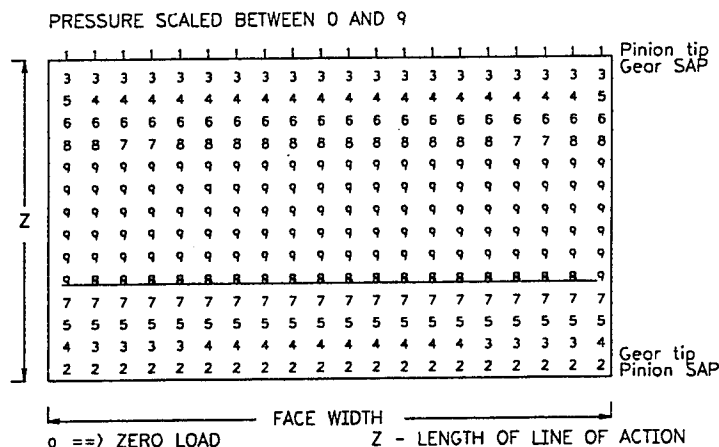Fig. 5. Profiles used in the LDP Simulation of the Baseline Spur Gear Pair

Fig. 6 Predicted Load Intensity Distribution for the Baseline Spur Gear Pair Operating at 2269 lbf-in.

pair, one sees in Fig. 8 that the minimum transmission error occurs at a much higher torque than the applied torques of 1361 and 2269 lbf-in., respectively. One of the actual profiles was digitized for this gear and it was found to give only a slightly different transmission error trace from that shown in Fig. 7. The differences between the digitized and the approximate profiles were always found to be less than the geometry variations that are encountered going from one tooth to the next.

Fig. 9 shows a load contour plot for the high contact ratio pair operating at 2269 lbf-in. In this case, one sees that the modification is such that contact does not reach the tip of either the pinion or the gear. The traditional interpretation
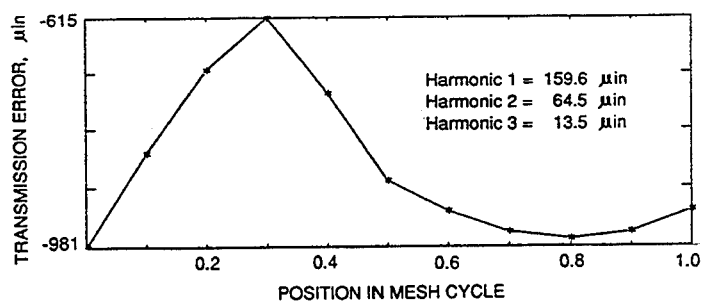
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Fig. 7 Predicted Transmission Error of the Baseline Spur Gear Pair Operating at 2269 lbf-in.
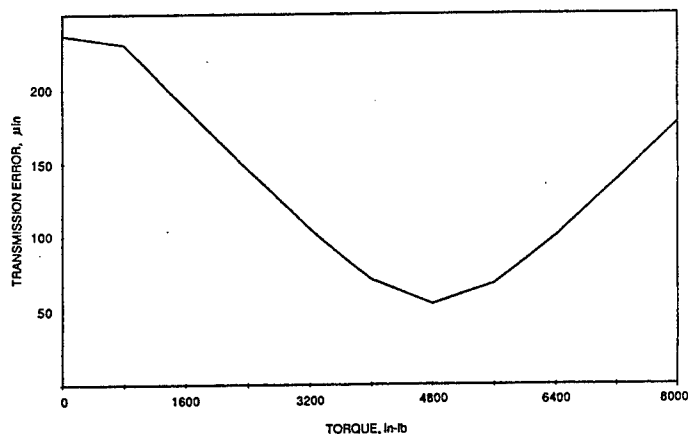
Harmonic 1 = 159.6 μin
Harmonic 2 = 64.5 μin
Harmonic 3 = 13.5 μin

Fig. 8. Effect of Torque on Mesh Frequency Harmonic of the Predicted Transmission Error of the Baseline Spur Gear Pair

PRESSURE SCALED BETWEEN 0 AND 9

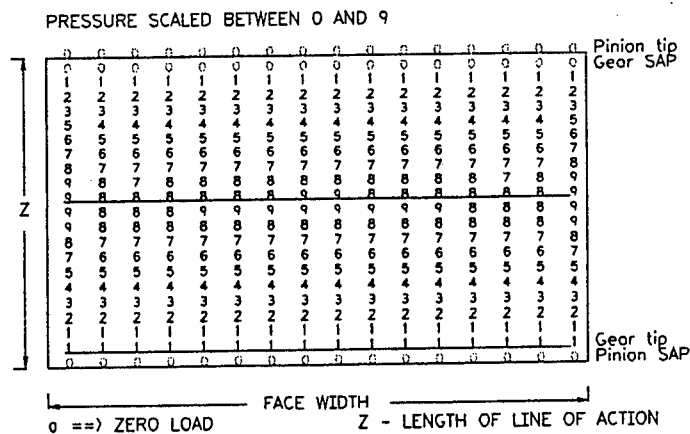0 ==) ZERO LOAD   Z - LENGTH OF LINE OF ACTION

Fig. 9. Predicted Load Intensity Distribution for the High Contact Ratio Spur Gear Pair Operating at 2269 lbf-in.

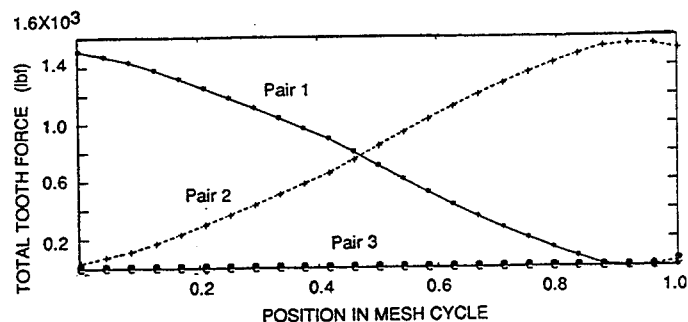Fig. 10. Predicted Mesh Cycle Forces Carried by High Contact Ratio Tooth Pairs Operating at 2269 lbf-in.

would be that the modification is slightly high for the gear set when operating at this load. In most of the other gear sets, the predicted contact patterns are close to those of Fig. 9. Similar contact patterns were observed in the actual testing.

Fig. 10 shows a different means of interpreting the load sharing between the high contact ratio spur gears. Here, the predicted force carried by three consecutive pairs of teeth through one mesh cycle are plotted. If one places end to end the traces of tooth pairs 3, 2, and 1, respectively, the tooth force trace for a single tooth would be created. The interpretation of this figure is that the modification is functioning very well - there are no discontinuities in the tooth forces and there are literally two pairs of teeth in contact throughout the mesh cycle giving an effective contact ratio of just under 2.0. Since a goal of high contact ratio teeth is to have a contact ratio of 2.0 or greater, this modification seems to work well. The fact that the contact of Fig. 9 does not reach the tooth tips is verified here, since the operating contact ratio is somewhat less than the theoretical contact ratio of 2.10.

Transmission Error Prediction/Measurement

Similar procedures were followed in modeling all 8 gear sets. The tricky exception is the double helical gear, since the standard LDP program does not have the capability of modeling this type of gear. However, in a previous study using a modification of LDP for double helical gears it was found that as long as each helix has the same profile and lead modifications, it is possible to model just half the mesh using one-half of the total load and the same results will be obtained [22].

Fig. 11 presents a bar chart of the predicted amplitudes of the mesh frequency transmission error at each harmonic for each of the test gear pairs. The decibel values are referenced to 0.1 micro-inch so that the predicted transmission errors at each harmonic range from close to 0.1 microinch for the third harmonic of gear pair 5 to over 300 micro-inches for the first harmonic of gear pair 7. In general, one sees that for each gear set, as the harmonic number increases, the amplitude of transmission error decreases. It was also found in the modeling process that the predictions of the higher harmonic amplitudes were much more sensitive to slight changes in the profile modifications than was the mesh frequency harmonic.

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When comparing individual gear sets, it is observed that in most instances, there is a good qualitative conclusion that as contact ratio increases, transmission error decreases. This is the same conclusion reached by other investigators [1,5] and also is similar to the noise measurement conclusion of Drago et al.[12]. In fact, if the gears are ranked in order of quietness for each harmonic, one finds that in most instances the rankings for transmission error of Fig. 11 are the same as the rankings for noise for Figs. 1-3.

One notable exception is the double helical gear pair (pair #4) which has a higher contact ratio than gear pair #5, the conventional helical gear set. Here, the predicted transmission errors are always slightly greater for the double helicals than for the regular helicals. Reasons for this discrepancy could be due to the reduction in the thrust load for the double helicals or due to the increased inertia of the double helicals due to the added material in the gap between the individual helices.

In almost every case, the transmission error predictions of the mesh frequency harmonic decrease when increasing the load from 1361 lbf-in to 2269 lbf-in. This is because each of the gear pairs has an "optimum" load curve similar to Fig. 8. and most of the gear pairs are operating to the left side of the curve. A major difference that was observed in these curves as contact ratio was increased is that the slopes on either side of the "optimum" are much less, indicating a lower load sensitivity as contact ratio increases. This reduced load sensitivity is one of the major advantages of increasing the contact ratio of gears. This load sensitivity trend is not as apparent for the higher harmonics of mesh frequency.

In order to be able to condense the comparis ons of the experiment and the predictions, we will use the scheme used by Drago, et al.[12], where they compared the noise of each of the gear pairs with the base-line spur gear pair (pair 1). This method of comparison should also normalize out any dynamic effects since the dynamics should be the same for each set of gears (the exception might be the double helical gears whose inertias are greater than the other gears. In this case, the first of the experimentally measured sound power levels of the base-line spur pair (pair #1a) of Figs. 1-3 was used for each harmonic's reference value.

Fig. 12 shows the comparison of the LDP predictions with the measured sound reductions of the first harmonic (mesh frequency) at 3000 rpm. For this condition, as with the subsequent conditions, data is always presented as a reduction in either transmission error or sound power with respect to the baseline spur gear set. This presentation method, which is logarithmic on each axis, essentially normalizes the results so that comparisons can be made without knowing the transmission error to noise coupling efficiency. Individual gear pairs are labeled on the plot, with the letter A indicating the lower torque and the letter C indicating the higher torque. It is felt that the 3000 rpm results should correlate best with the static predictions of LDP since the lowest speed should have the lowest amount of dynamics.

The correlation in Fig. 12 is roughly linear and the major deviations from the straight line correlation are the high contact ratio spur gear pair (pair #2) and the double helical pair (pair #4). The trends are similar but with more scatter for the comparisons at 4000 rpm and 5000 rpm that are respectively presented in Figs. 13 and 14. Second harmonic comparisons for all speeds are presented in Fig. 15 and third harmonic data are presented in Fig. 16. It appears that as the harmonic number goes up, there is more scatter in the comparison. This might be expected, since higher harmonics are affected more by minute changes that could be encountered in modeling the profiles or in changes in the test setup.

Even though transmission error is purported to be closely allied with noise level, the data does not seem to show a universal constant that correlates the two values. This constant, which is obtained from the slope of the best straight line fit of the sound power vs transmission error curve, is 0.56 for 3000 rpm, 0.37 for 4000 rpm and 0.55 for the 5000 rpm data. Because of system sensitivities such as system dynamics, force transmission paths and housing characteristics one would expect this constant to change from one system to the next and from one frequency to the next. Also, the effect of helix angle on the force coupling between the shaft and the housing might have a significant effect on this constant.

In Figs. 12-16, each data point is affected by many factors which could cause the point to have a different location on the plot. These factors include measurement variation and measurement error on the vertical axis and modeling variability on the horizontal axis. A simple analysis of measurement variability can be made from Figs. 1-3 which show a 2-4 dB sound power variation due to simply remounting the same set of spur gears. Since this gear set is used as the reference for the plots, any errors in measurement or modeling will affect the "zero" position of Figs 12-16. However, this change would not significantly affect the slope constants mentioned in the previous paragraph. There are also issues of accuracy of the error charts that were used in LDP, mounting misalignment, errors in modeling of tooth compliance, etc., that should provide horizontal spread in each data point. These issues could be studied with the model in order to determine each gear pair's sensitivity to these factors.

Summary

This paper has shown that at any one operating condition and measurement frequency there is a reasonable correlation between predicted transmission error and measured sound power levels. This correlation is better for the mesh frequency harmonic and decreases as the harmonic number is increased. Many of the variations are well within what are projected to be variations of profiles from tooth to tooth, modeling errors, tooth measurement errors and testing variations. These issues will be addressed in future studies of these data. Also, the seemingly high prediction of noise
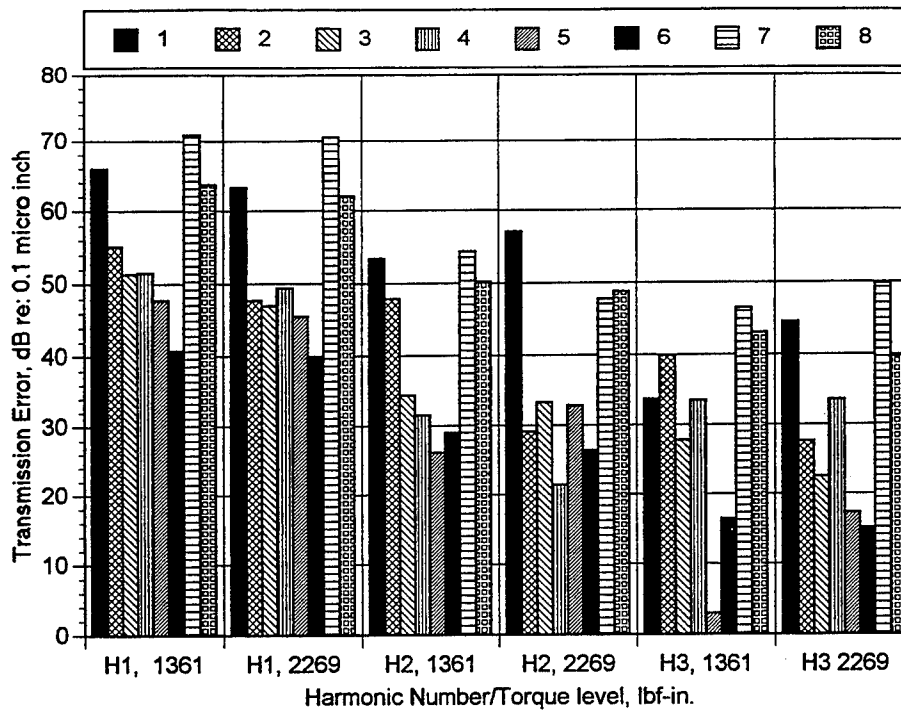
6

Fig. 11. Summary Chart of Predicted Transmission Errors for Each of the First Three Mesh Harmonics at both 1361 and 2269 lbf-in.
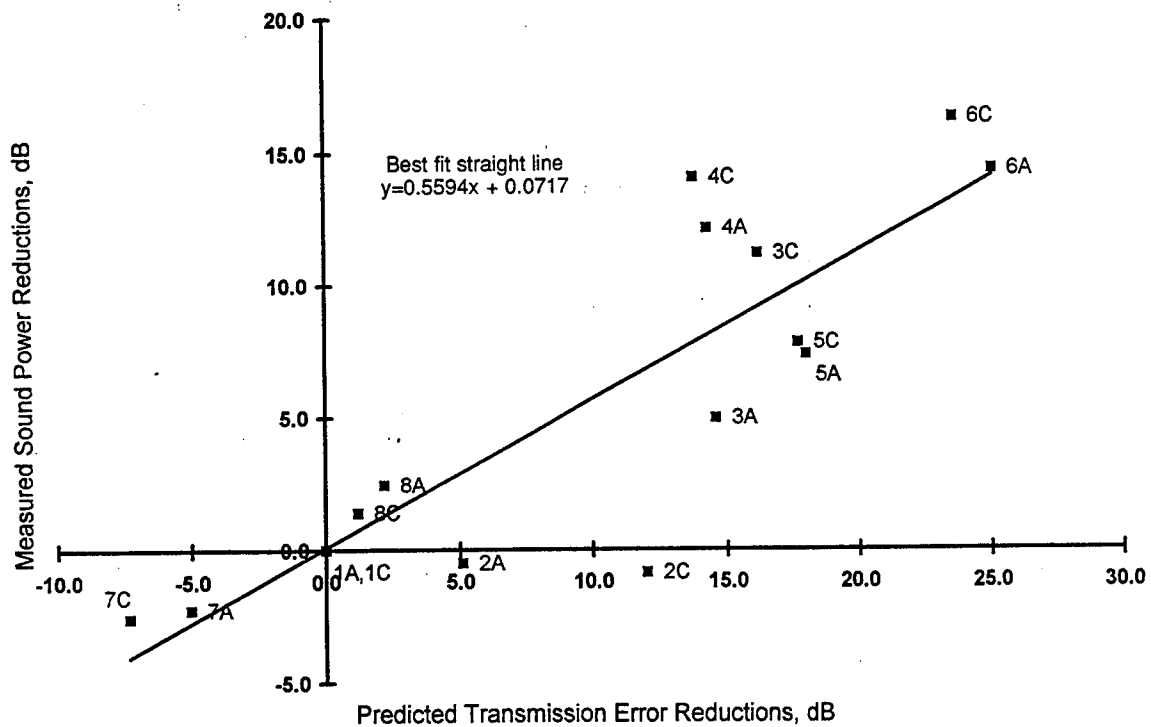

Fig. 12. Comparison of Predicted Transmission Error with Measured Sound Power Levels for the Mesh Frequency Harmonic at 3000 rpm (Data Reported as Reductions from Baseline Spur Gear set).
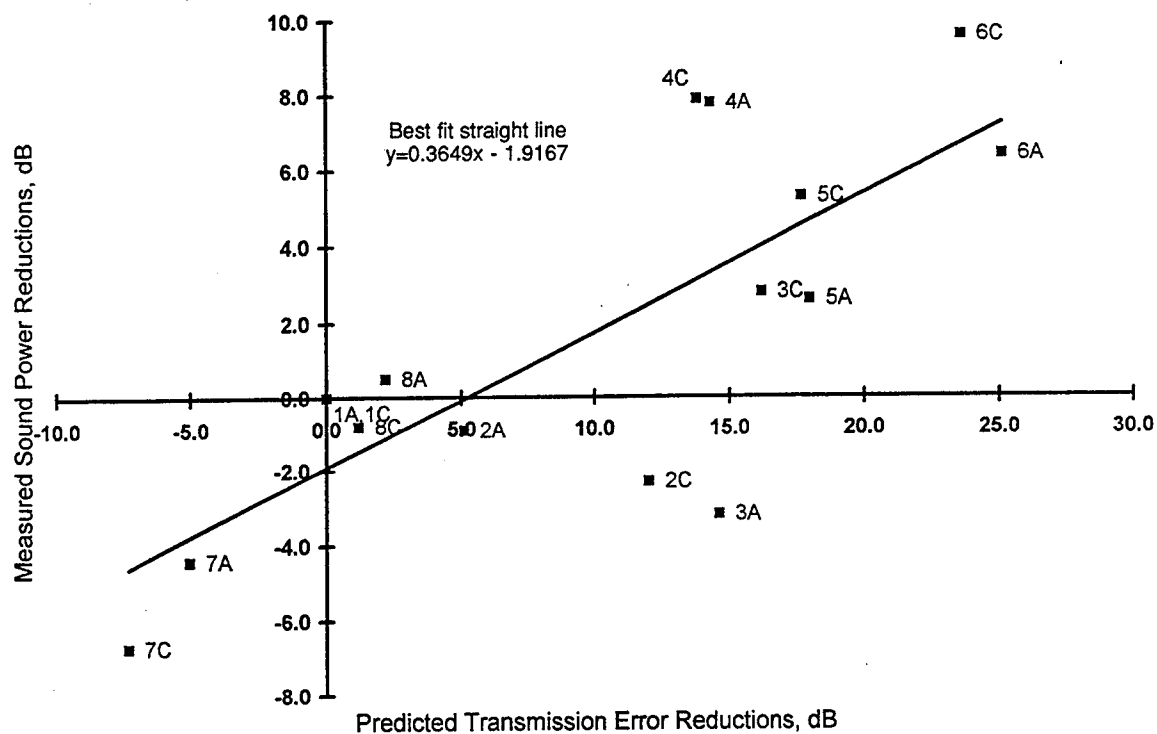
Fig. 13. Comparison of Predicted Transmission Error with Measured Sound Power Levels for the Mesh Frequency Harmonic at 4000 rpm (Data Reported as Reductions from Baseline Spur Gear set).
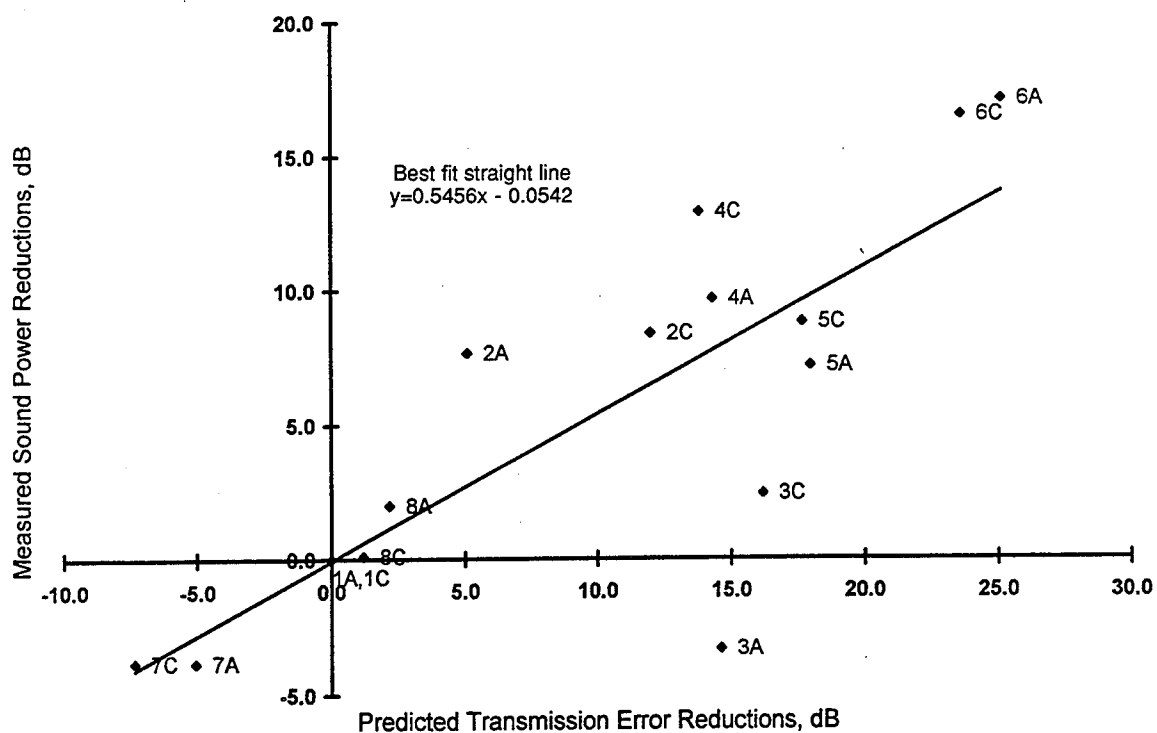

Fig. 14. Comparison of Predicted Transmission Error with Measured Sound Power Levels for the Mesh Frequency Harmonic at 5000 rpm (Data Reported as Reductions from Baseline Spur Gear set).
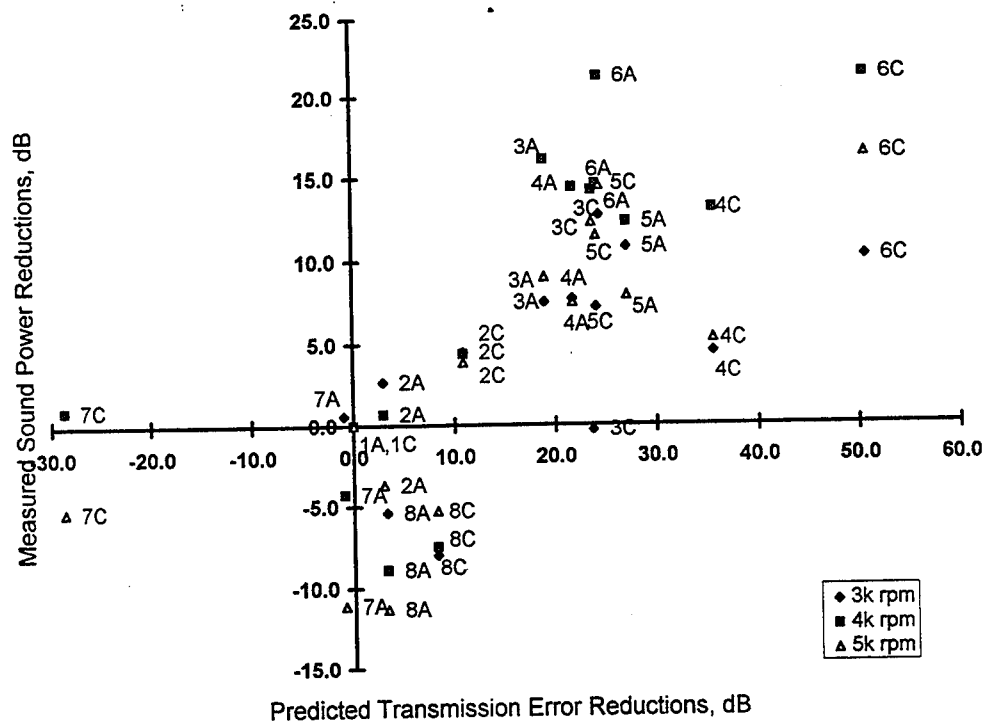
Fig. 15. Comparison of Predicted Transmission Error with Measured Sound Power Levels for the Second Harmonic of Mesh Frequency (Data Reported as Reductions from Baseline Spur Gear set).
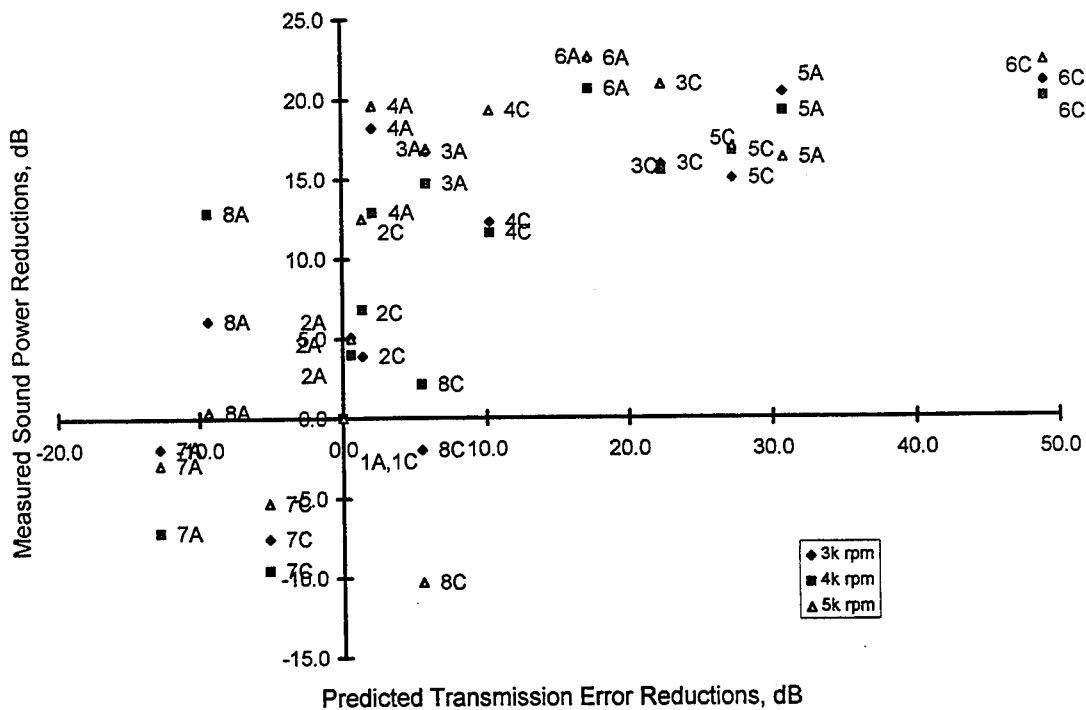

Fig. 16. Comparison of Predicted Transmission Error with Measured Sound Power Levels for the Third Harmonic of Mesh Frequency (Data Reported as Reductions from Baseline Spur Gear set).

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reduction for the high contact ratio spur gear pair needs to be studied in more detail. Perhaps other noise generation sources in the high contact ratio spur gears cancel out the transmission error gains. Finally, additional gears should be cut that have modifications that are optimized for the test load. In some of the simulations that were run, further noise reductions of as high as 20 dB below the "best" reductions of gear pair 6 were predicted using LDP. Since other excitation sources [1] are likely to start dominating after the primary excitation is nearly eliminated, it is not expected that the measured sound power reduction will be as great as the predicted transmission error reduction for the optimally designed gears.

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13. ABSTRACT (Maximum 200 words)

Measured sound power data from eight different spur, single and double helical gear designs are compared with predictions of transmission error by the Load Distribution Program. The sound power data was taken from the recent Army-funded Advanced Rotorcraft Transmission project. Tests were conducted in the NASA gear noise rig. Results of both test data and transmission error predictions are made for each harmonic of mesh frequency at several operating conditions. In general, the transmission error predictions compare favorably with the measured noise levels.

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